Invest igation of Gas Effects on Cryocooler Resonance Characteristics

M. K. Heun, S. A. Collins, D. L. Johnson, and R. G. Ross Jr.

Jet Propulsion Laboratory
California Institute of Technology
Pasadena. CA 91109

ABSTRACT

Cryocooler thermal and vibrational performance is determined, fundamentally, by the dynamic interactions between the mechanical system and the working fluid. "l'his paper explores the effect of working-fluid characteristics on the mechanical response of the cooler. Experimental data collected from two coolers characterized under the Jet Propulsion Laboratory's extensive program of cryocooler testing and characterization show that a classical single-degree-of-freedom spring-mass-damper model does not capture the full frequency dependence of the mechanical response. The data from two modes of cooler operation (slosh and head-to-head) are used to motivate the explanation that working-fluid characteristics dominate at high frequencies, and mechanical system characteristics dominate at lower frequencies. Operating temperature is shown to be a significant factor in determining resonance behavior. Finally, the discussion provides a framework within which resonant parameters and cooler characteristics can be extracted from the experimental data.

INTRODUCTION

The growing demand for long-wavelength infrared imaging instruments for space observational applications has led to the ongoing development of cryocoolers, mechanical refrigerators which typically provide cooling at 30–100 K with refrigeration loads of 1-10 W.' Because these units are used to cool imaging system detectors, both the vibration of the coldhead assembly and the force transmitted to the structure are important for image integrity and overall instrument performance. And, characterization of cryocooler vibration and force signatures is an essential element of instrument modeling and design, Thus, an understanding of the parameters that affect coldhead vibration and transmitted force is required for optimized instrument design.

A previous publication highlighted the theory behind cryocooler resonance characterization for single-piston Stirling coolers, focusing, in particular, on launch vibration response, The present paper extends the previous results by examining the resonance characteristics of a "back-to-back" cooler and identifying the helium working fluid as an important element affecting cooler dynamics.

Cooler Designs

Fig. 1 a shows a schematic diagram of the compressor motor of a typical single-piston Stirling cycle cryocooler, the subject of the earlier study.[2] Fig. lb shows the back-to-back unit evaluated in the present paper. Hack-to-back systems are said to be operating in "head-to-head" mode when the two pistons move toward the center simultaneously. Refrigeration is achieved only during head-to-head mode operation. For testing purposes, a back-to-back system may be driven in "slosh" mode: as one piston moves toward the center, the other piston moves toward the end. The slosh and head-to-head modes are illustrated in Fig. 2.

In head-to-head mode (normal operation), the compression work of the pistons on the helium is necessary to obtain refrigeration. Mechanically, the gas acts as a spring which compresses during volume reduction, and it provides darnping as the gas is forced through the transfer tube, the coldhead, and the regenerator matrix. In slosh mode, no compression or expansion of the gas occurs. In effect, the gas spring and damper arc eliminated from the system, leaving only the mechanical spring and its dynamic characteristics.

Dynamic Models

The motion of the moving masses (pistons) may be described by the classical spring-mass-damper equation:[3]

$$m\ddot{x} + c\dot{x} + kx = F_d \sin(2\pi f t) \tag{1}$$

where $F_d = K_{mf}\, i$ is the amplitude of the drive force, K_{mf} is the motor force constant, and i is the zero-to-peak current supplied to the motor. This model is hereafter termed the "classical model," and the system which it describes is said to be the "classical system, "

The zero-to-peak stroke response x of the classical system to the sinusoidal excitation at frequency f is given by

$$\frac{x}{x_o} = \frac{1}{\sqrt{\left[1-\left(\frac{f}{f_o}\right)^2\right]^2 + \left[2\zeta\left(\frac{f}{f_o}\right)\right]^2}}, \tag{2}$$

where $x_o = F_d/k$ is the static displacement, $\zeta = c/c_c$ is the damping ratio, $c_c = 2(km)^{1/2}$ is the critical damping coefficient, and $f_o = (k/m)^{1/2}/(2\pi)$ is the undamped natural frequency. The phase ϕ of the stroke relative to the applied force is given by

$$\tan\phi = \frac{2\zeta\left(\frac{f}{f_o}\right)}{1-\left(\frac{f}{f_o}\right)^2}. \tag{3}$$

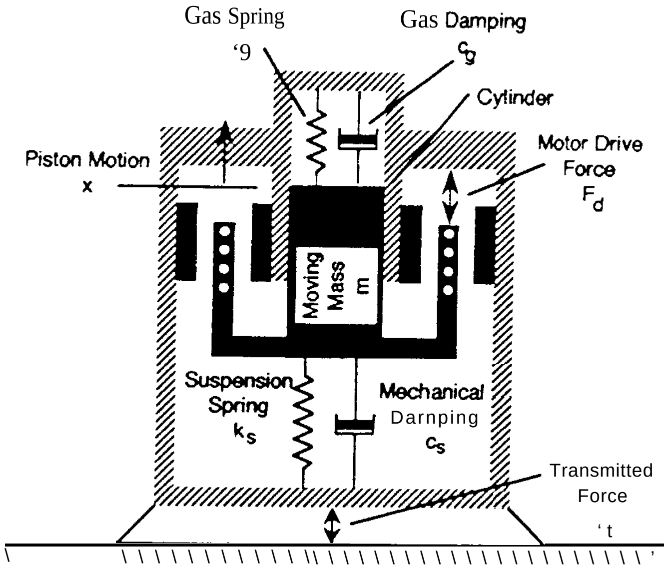
The force transmitted to the base F_t is given by

$$\frac{F_t}{F_d} = \frac{\left(\frac{f}{f_o}\right)^2}{\sqrt{\left[1-\left(\frac{f}{f_o}\right)^2\right]^2 + \left[2\zeta\left(\frac{f}{f_o}\right)\right]^2}}, \tag{4}$$

and the phase of the transmitted force relative to the applied force is identical to Eq.(3).

Experimental Procedures

During laboratory testing, a sinusoidal drive current is applied to the cooler motors. The piston stroke, transmitted force, and the phases of both stroke and transmitted force relative to the drive current are measured. The above data is collected as a function of drive frequency f, drive current amplitude i, and coldblock temperature. The effective spring stiffness k_s, the



a. Single-piston cooler.

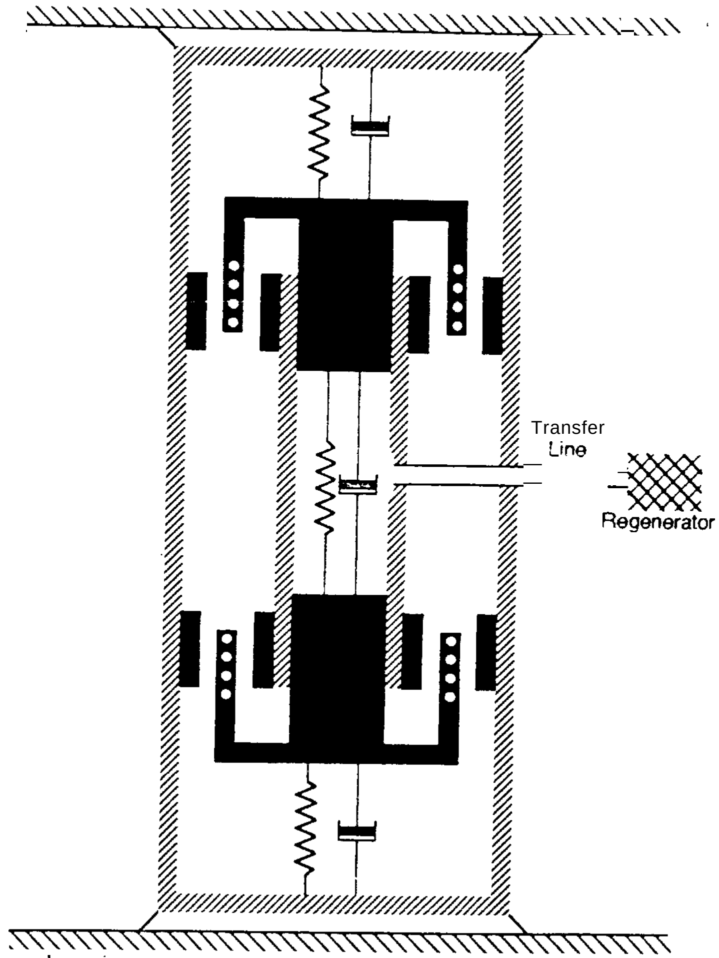


Figure 1. Cooler motor schematic.

b. Back-to-back cooler.

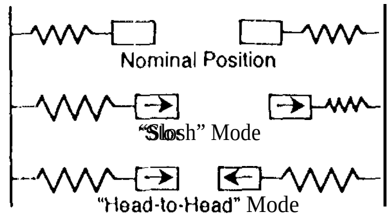


Figure 2. Modes of piston motion.

effective damping coefficient c, the motor force constant K_{mf}, and the moving mass m are obtained by curve fitting the classical model to the slosh mode data.

Because the piston inertia cancels internally, head-to-head mode operation results in negligible force transmitted to the supporting structure. In fact, the magnitude of the transmitted force was below the sensitivity of our instrumentation during head-to-head mode operation, During slosh mode, however, significant and measurable transmitted forces were observed. Low frequency (< 10 Hz) slosh mode operation resulted in transmitted forces that were below the level of the ambient background noise. Although we present these low-frequency data, they were neglected during the curve-fitting procedure.

Experimental measurements from one of the back-to-back coolers evaluated under JPL's test program are presented herein. Data from a second cooler showed similar results and are not presented here.

EXPERIMENTAL RESULTS

Slosh mode

Fig. 3 shows stroke amplitude and phase vs. frequency for slosh mode operation at two drive current levels. The solid lines show the classical model curve tit, and the data points are experimental measurements. The cooler response conforms to the classic model, and the natural frequency of the system is about 24 Hz. The natural frequency exhibits a minor sensitivity to the drive current, and the effective spring stiffness increases slightly with amplitude.

In slosh mode, no gas compression occurs between the pistons, and the parameters derived from the slosh mode curve tit reflect the characteristics of the mechanical springs themselves. Thus, the mechanical springs are found to have a stiffness k_s = 8 N/mm, a damping ratio c/c_c = 0.010 and a motor force constant K_{mf} = 22 N/A.

For completeness, Fig. 4 shows the transmitted force amplitude and phase vs. frequency. Again, the data follow the classical model presented above.

Head-to-head mode

Cooler Response. Fig. 5 shows both the amplitude and the phase of the stroke relative to the drive current. Two classical models are presented on the graphs, The solid line is a classical model response generated with the parameters (k_s, c/c_c, and K_{mf}) derived from the slosh mode testing. Or, put another way, the solid line shows the predicted response of the system assuming no gas participation in the overall system stiffness or damping, We see that at low frequencies (< 10 Hz) the classical model nearly matches the experimental data, indicating that the parameters of the mechanical spring are sufficient to describe the system response in this frequency regime.

We see, however, that the resonant frequency (the frequency of maximum forced amplitude) is higher for head-to-head mode operation than slosh mode operation, This indicates a stiffer system in head-to-head mode operation. Furthermore, the response curve is flatter near resonance, indicating increased damping. The dashed line shows a curve tit to the data

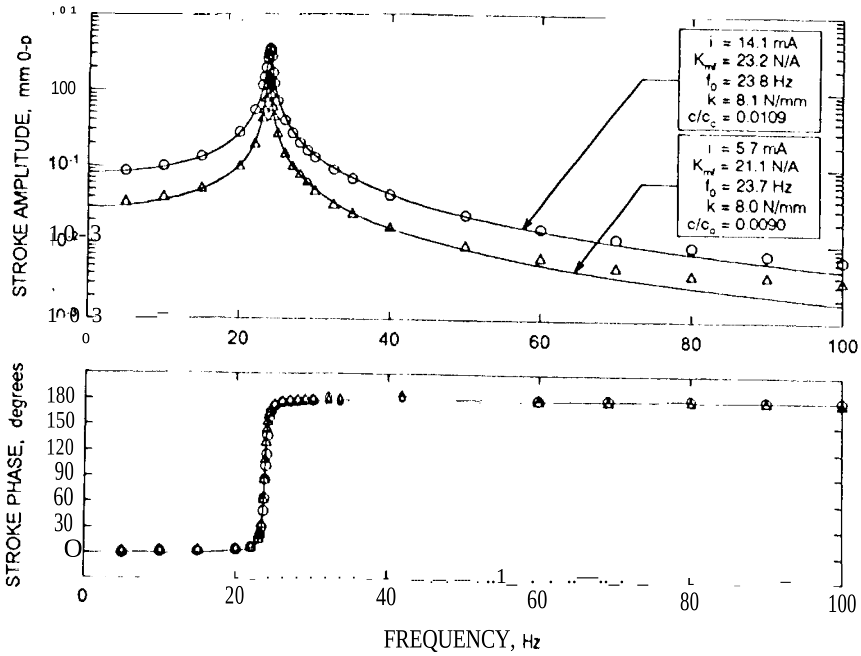


Figure 3. Slosh mode stroke.

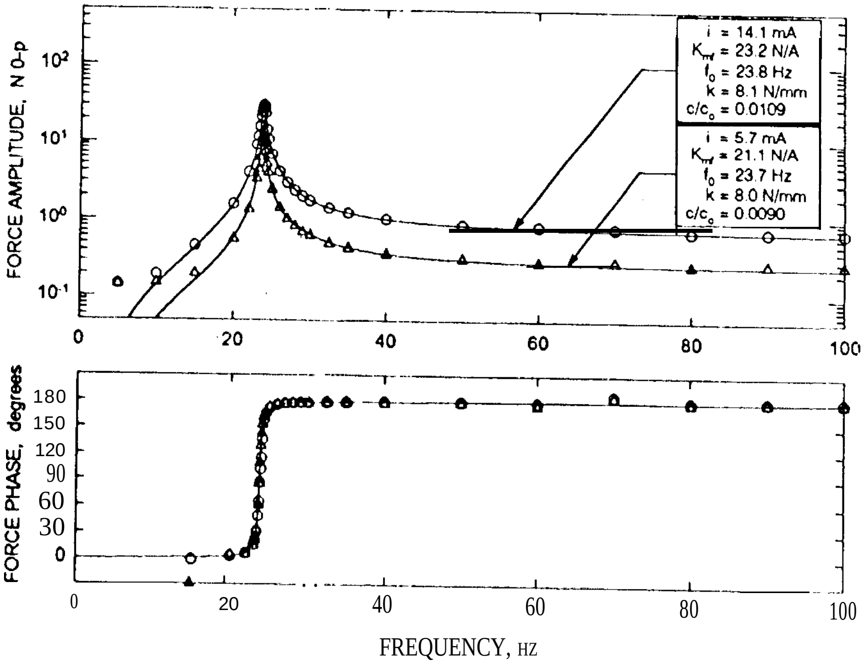


Figure 4. Slosh mode force.

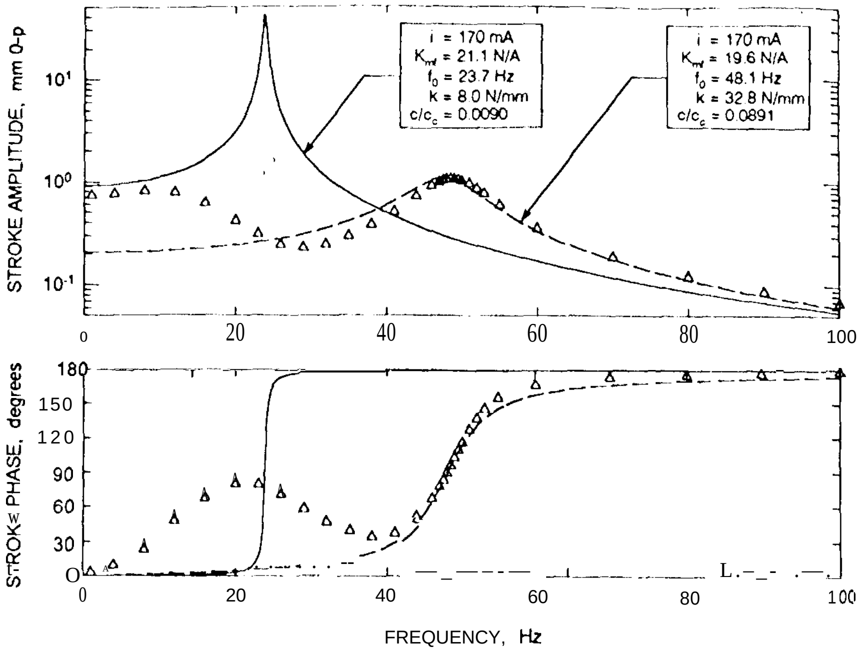


Figure 5. Head-to-head mode stroke, ambient temperature,

above 40 Hz. The high frequency data can be represented by a classical system with a stiffer spring and more damping compared to the low frequency data.

We postulate that at low frequency, gas effects are minimal, Gas flows readily through the regenerator, through the coldhead, and between the piston and the cylinder, and little gas compression occurs. At high frequency, gas compression is significant as no gas slips between the piston and the cylinder wall, and the regenerator provides significant flow resistance: the gas acts both as a spring and a damper. Thus, at low frequency, the cooler response is dominated by the characteristics of the mechanical spring, and the system is said to be mechanically dominated. At high frequency, the cooler response is controlled by the characteristics of the gas, and the system is said to be pneumatically dominated.

Fig. 5 shows a transition region (10 Hz < f < 40 Hz), wherein the amplitude shifts from the mechanically dominated regime to pneumatically dominated behavior. We see also that the phase exhibits a transition between the two regimes. As expected, at low frequency the phase approaches 0°, at high frequency it approaches 180°. The head-to-head mode damped natural frequency (defined to be the frequency where the phase is 90°) is 48 Hz. However, between the mechanically dominated regime and the pneumatically dominated regime, the phase exhibits a peculiar transition. One expects a rather gradual transition from the phase response of the mechanically dominated system (shown by the solid line) to the response of the pneumatically dominated system (shown by the dashed line). Instead, Fig. 5 shows that a complex interaction exists between the two regimes. This effect is manifest as a wave on the phase plot, and it is termed the "transition phase lead" because the measured phase "leads" the classical model as frequency increases,

The implications of the observed behavior on cooler vibration control systems are significant, Although coolers are designed to work only at the head-to-head mode resonant frequency (in this case, f = 48 Hz), off-design capabilities are an essential aspect of a robust control algorithm. It is clear that such a robust algorithm should not expect classical system

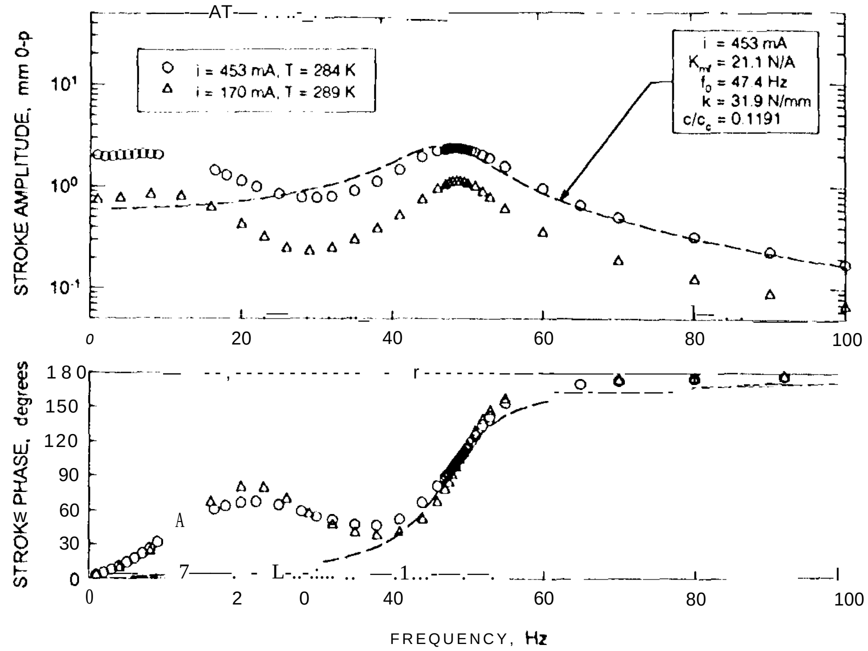


Figure 6. Effect of drive current on head-to-head mode stroke, ambient temperature.

behavior at frequencies below the operating frequency, particularly in terms of the phase response. Near-resonance phase control and dynamic modeling are significantly impacted by gas effects.

Effect of Drive Current, Ambient Conditions. Fig. 6 shows the sensitivity of cooler response to drive current. As expected, the higher drive current generates larger stroke. The resonant frequency is not significantly altered by the increased stroke, although the phase and the values of c/c, indicate a positive correlation between damping and amplitude. The larger stroke results in a more highly **damped** response, likely due to the difference in the dynamic behavior of the gas. The amplitude of the wave on the phase plot decreases as amplitude increases,

Effect of Operating Temperature. Fig. 7 shows the effect of operating temperature on the amplitude and phase of the system. The data represented by the triangles was collected with the coldhead at about 290 K (ambient conditions) whereas the data represented by the diamonds was collected with the coldhead at cryogenic temperatures (about 45 K). In each case, the motor drive current was selected to provide identical peak stroke (about 1 mm_{0-p}) at the resonant frequency (48 Hz). Clearly, operating temperature significantly affects the dynamic behavior of the cooler, and there are many differences between cryogenic and ambient operation.

Looking first at the amplitude data, we see that cryogenic operation requires substantially higher motor current (283 mA) compared to ambient conditions (170 mA) to achieve the same stroke at resonance. Relative to ambient operation, the resonant peak in the cryogenic data is broader than the corresponding peak in the ambient data, indicating a more highly damped system under cryogenic conditions, The higher darnping is verified by the classical model tit to the high frequency data: damping increases significantly while the overall stiffness and the motor force constant are relatively unchanged when moving from ambient to cryogenic operation, A likely explanation for the increased damping is the increased density of the helium gas in the cold-block at cryogenic temperatures.[1] Once again, the gas characteristics are seen to impact significantly the resonant behavior of the coolers.

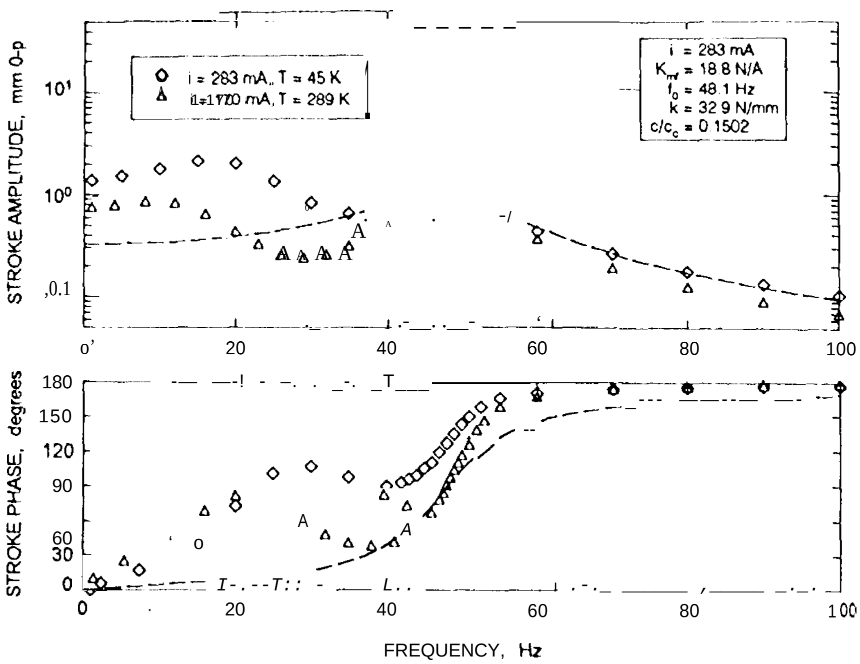


Figure 7. Effect of coldhead temperature on head-to-head mode stroke,

The dashed line in Fig. 7 is a curve fit of the classical model to the amplitude data above 40 Hz. Similar to the ambient data in Fig. 5, the classical model tits the amplitude data nicely. However, unlike the ambient data in Fig, 5, the classical model poorly represents the cryogenic phase data. The transition phase lead for the cryogenic data is higher in frequency and smaller in amplitude compared to the ambient data. Surprisingly, the phase crosses 90° three times. Near the resonant frequency, we observe a 90° crossing at 39.8 Hz and again at 40.2 Hz. Finally, a 90° crossing occurs at 23.0 Hz. Figure 7 shows that the phase behavior is a function of both the operating temperature and stroke amplitude. Clearly, the classical model does not capture the behavior of the system.

The cryogenic data shown in Fig. 7 amplify the conclusions drawn from the ambient data shown in Fig. 5, namely that active vibration control strategies must include gas effects if a complete model of the cooler dynamic behavior is required. This is more important with phase than with amplitude. The cryogenic data shows that the amplitude can be represented by a classical model, but that the, phase behavior cannot be captured by parameters derived from the stroke data.

Effect of Drive Current, Cryogenic Conditions. Fig. 8 shows the impact of motor drive current at cryogenic conditions. Similar to Fig. 6, we see that higher drive current generates larger stroke. The lower drive current generates a larger amplitude transition phase lead. At both cryogenic and ambient operating conditions, higher drive current results in a more highly damped system. The dashed line represents a classical model curve fit to the high drive current data at frequencies higher than 40 Hz. As shown in the previous section, the classical model fails to capture both the stoke and phase response simultaneously.

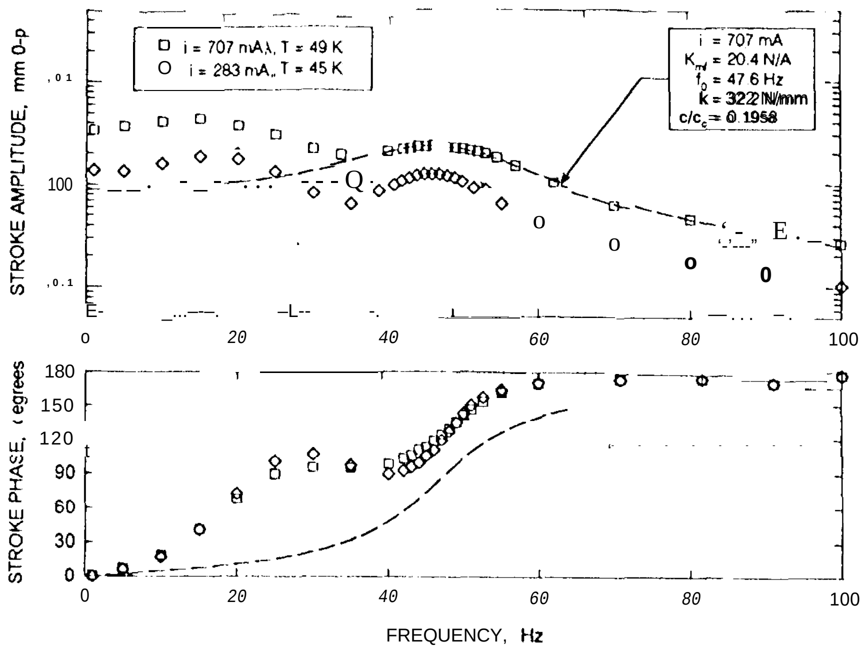


Figure 8. Effect of drive current on head-to-head mode stroke, cryogenic temperature.

CONCLUSION

Previous papers on cryocooler resonance characteristics have neglected phase information when characterizing cooler parameters, in part because the previous studies focused on cooler response to launch-induced vibration. In this paper, we have examined both the amplitude and phase response of back-to-back cryocoolers to find that pneumatic effects have a significant impact on cooler resonance characteristics.

Three regimes were observed during head-to-head mode operation, (1) The mechanically dominated regime occurs at low frequencies where gas effects are minimized and the characteristics of the mechanical spring control the resonant behavior of the cooler. (2) The pneumatically dominated regime occurs at high frequencies where the gas characteristics predominate. (3) Between the mechanically dominated regime and the pneumatically dominated regime, a transition regime exists that is nearly 30 Hz in extent.

We showed that the classical model is insufficient to describe head-to-head piston motion across a broad range of frequency, At ambient conditions, two classical models may be used to describe both the amplitude and phase response of the coolers. A different set of parameters is required for each regime. At cryogenic conditions, a classical model may be employed to describe the amplitude but not the phase in the pneumatically-dominated regime. In the transition regime, the amplitude makes a smooth shift from the mechanically-dominated regime to the pneumatically-dominated regime, The phase exhibits peculiar behavior that is termed the "transition phase lead," Further work should be undertaken to describe completely the cooler behavior in the transition regime. The phase characteristics merit particular attention because of the importance of phase information for active vibration control algorithms.

Ultimately, a model that completely captures gas effects on cryocooler resonance may be used in conjunction with diagnostic testing. The results may allow determination of parameters that influence gas behavior such as the clearance between the piston and the cylinder, the piston length and diameter, and the effective flow resistance of the transfer tube, coldhead, and regenerator.

ACKNOWLEDGEMENTS

The work described in this paper was carried out by the Jet Propulsion Laboratory, California Institute of I'ethnology, and was sponsored by the NASA EOS AIRS project, under a contract with the National Aeronautics and Space Administration.

REFERENCES

1. Ross, R. G. Jr., "Requirements for long-life Mechanical Cryocoolers for Space Applications", *Cryogenics*, vol. 30, (1990), pp. 233-238.

2. Ross, R. G. Jr, et al., "Cryocooler Resonance Characterization'', *Cryogenics*, vol. 34, no. 5 (1994), pp. 435-442.

3. Thomson, W.T., *Theory of Vibration with* Applications, 3rd ed., Prentice Hall, Englewood Cliffs, New Jersey, (1988), pp.50-81.